THE UNIVERSITY OF ALBERTA

DECISION MODEL FOR OVERLAND WATER PIPELINES IN NORTHERN CANADA

by

© GARRY DUANE LAVOLD

A THESIS

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ABSTRACT

Providing adequate water, sewerage and garbage facilities for the small and isolated communities of northern Canada is a major problem of the Government of the Northwest Territories. Working towards a solution of this problem, the Territorial Government proposed a multi-million dollar, ten year program designed to improve the water and sanitation services to the some sixty scattered communities under its jurisdiction.

Present studies for this undertaking have focused on the overall scope of the project. The next phase is to narrow the scope and to evaluate alternative solutions, with enough detail, to ascertain the most pragmatic and economic solution for each phase of the project. One area of concern is the movement of water between a source and a community. In southern Canada, the primary method of transporting water to communities is by a pump and buried pipeline system. To use this type of system in the Northwest Territories presents special problems, as most areas are in the permafrost zones, making a buried line usually unfeasible. Thus, present pipeline costing formulations are generally not usable for these systems in the North. This thesis presents a mathematical model for northern pipeline systems to determine physically possible systems and the present value costs of the different systems.

The model developed provides for alternative types of pump and pipeline systems--with a strong emphasis on various methods of preventing line freeze up (tracing the line, insulation, preheating the source water). The effects of the various methods of line freeze-up

prevention will be illustrated by generating numerical examples with the model. The model will allow for the comparison of the various alternatives on a net present value basis, including both capital costs and associated operating expenditures.

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INTRODUCTION

Problem Statement

A major problem faced by the Government of the Northwest Territories is to upgrade the water and sanitation facilities of the communities under its jurisdiction. At present, most communities within the Northwest Territories are operating under primitive water and sanitation facilities. The Territorial Government has presented a proposed policy statement which recognizes this need, and which outlines a multi-million dollar program to improve water and sanitation systems. Part of this improvement will involve the determination of a method of transporting water to each community. The common system of transporting water in southern Canada is by underground pipelines. Because most of the Northwest Territories is in the permafrost zone, water pipelines for the area would need to be overground. This thesis considers the problem of transporting water by an overground pipeline in a severe northern climate.

The construction of an overland pipeline for water transportation in a very cold climate requires a careful consideration of line freeze up prevention. This problem of line freeze up prevention controls the design of the total system's components. Determining the effects of one individual variable in a system of closely interrelated variables presents a complex problem. The main input variables to be considered are these: (1) the water source temperature, (2) the water quantity required by the community, (3) the distance between the

water source and the community, and (4) the lowest ambient temperature and related wind velocity.

The main physical parameters considered are: (1) the velocity of water in the pipe, (2) insulation thickness, (3) heat transfer between the surroundings and the water, (4) the diameter of the pipe, and (5) the pump size.

The above parameters are all interrelated. Thus, the problem considered involves, firstly, the analysis of a group of interrelated input variables and design parameters to define the physical characteristics of water moving in an insulated pipe which is located in an extremely cold environment.

The second part of the problem is to provide a standard of comparison for the alternative overland systems which meet the physical constraints. This is done by providing capital and operating cost equations for the different alternatives. The physical systems are evaluated using the net present value of their associated costs. Normally, the system with the lowest net present value will be recommended for a given community.

The chapters of this paper are arranged as follows. Chapter II presents the detailed qualitative development of the required physical equations. Chapter III gives an example using the equations to determine which terms are significant. Chapter IV presents a group of simplified physical equations with sample calculations. Following the examples is the cost equation development using the simplified equations. At this point the evaluation model is completed. Chapter V presents the conclusions.

Component Description

The pipeline system has three main components which can be utilized in different configurations to supply the water without its freezing. They include (1) pump and associated driver, (2) pipeline, and (3) a freeze up prevention system.

The pump chosen for all applications is a centrifugal pump. This type of pump has a proven reliability record, is simple in design and operation, and is readily available at a reasonable cost. Compared to the centrifugal pump, a positive displacement pump is usually more troublesome and requires more maintenance and care in operation. The pump driver can be electrical, gas turbine, diesel or steam turbine. The type of driver will, of course, depend on the availability of an energy source and cost. For most cases the driver will probably be an electric motor.

The pipeline considered will be standard schedule 40 steel pipe. The pipe sizes employed will be rounded up to standard size so that all evaluations are made using a pipe size that is readily available. However, the user may insert another size without modifying the model.

Of prime importance is the thickness and type of insulation required to prevent freeze up. Associated with this is the temperature of the water source and its related heat transfer. Variations in insulation thickness, pipe size and water source temperature must all be considered in evaluating the component configuration.

Qualitative Overview of Equation Development

To solve the above problem, a detailed mathematical model is developed. The model has two distinct parts. The first part contains

the equations for the physical characteristics of the system. The second part gives the associated cost equations.

The component equations will be sequentially arranged to consider the following: (1) water quantity, (2) water velocity, (3) pipe size, (4) convective heat transfer from the ambient conditions into the flowing water, (5) heat required to maintain a constant heat flux per unit pipe length, and (6) the distance water can travel without freezing, with a given set of components and input variables.

The cost equations then give comparative costs for each physical system.

CHAPTER II

PHYSICAL EQUATION DEVELOPMENT

The equation development will center on the necessary heat transfer considerations because a change in any of the input variables affects the possibility of line freeze up. However, the dimensions of the pipe, the water velocity and line pressure drop must be considered first.

Fluid Flow Section

First the size of the pipe must be considered. The inside diameter of the pipe is found with Equation (1)[1] which is

$$D = ((0.408 \cdot Q) \cdot VW^{-1})^{0.5} \qquad (1)$$

where the nomenclature is defined as follows:

D : internal diameter of pipe, in inches

Q : rate of flow, in U.S. gallons per minute

VW: mean velocity of flow of water in the pipe in feet per second

0.408: a constant to maintain unit consistency

The value of Q is characteristic of each community, so it would be specified for each case. It may also be varied to study the effects of larger or smaller systems, allowing for different anticipated growth levels. The velocity of the water is an input variable which gives the model an initial value. It will normally be in the range of 7 to 10 feet per second. The pipe diameter found should be rounded to the next larger commercial pipe size. For schedule 40 pipe, in a range of sizes likely to be encountered, the following table can be used:[2] (Table I)

TABLE I

COMMERCIAL WROUGHT STEEL PIPE DATA

SCHEDULE WALL THICKNESS--PER ASA B36.10-1950

(For Schedule 40 Pipe)

Nominal Pipe Size (Inches)	Outside Diameter (Inches) OD	Thickness (Inches)	Inside Diameter (Inches) ID
1	1.315	0.133	1.049
1½	1.900	0.145	1.610
2	2.375	0.154	2.067
3	3.500	0.216	3.068
4	4.500	0.237	4.026
5	5.563	0.258	5.047
6	6.625	0.280	6.065
8	8.625	0.322	7.981
10	10.75	0.365	10.02
12	12.75	0.406	11.938
14	14.0	0.438	13.124
16	16.0	0.500	15.000
18	18.0	0.562	16.876
20	20.0	0.593	18.814
24	24.0	0.687	22.626

The use of Table I should be adequate for most design studies, but if the user wanted a different pipe, he could input the pipe internal diameter after solving Equation (1). Note that this change in diameter will require the re-evaluation of the water velocity (VW). This is done as required in subsequent equations.

The next item is the amount of pressure drop that occurs in the line. To determine this, the actual line inside diameter (ID from the preceding table) should be used. For example, if D calculated was 2.2 inches, then ID for the next larger commercial size is 3.068 inches. Using this, the line pressure drop per hundred feet of pipe can be expressed as follows:[3]

$$f = 0.2083 \left(\frac{100}{C}\right)^{1.85} \frac{Q^{1.85}}{(ID)^{4.8655}} \tag{2}$$

where

f : friction head in feet of liquid per 100 feet of pipe

C : constant accounting for surface roughness (commonly used value for design purposes of welded steel pipe is 100)

ID: inside diameter of pipe, in inches

0.2083: a constant to maintain unit consistency

Equation (2) gives the line pressure drop per 100 feet of pipe, so for a source a given distance from the community, the overall line pressure drop is given by Equation (3)

$$LPD = f \cdot 0.433 \cdot sp.gr \cdot DS/100 \tag{3}$$

where

LPD : is the line pressure drop in pounds per square inch

sp.gr: is the specific gravity of the liquid, which for water is one

DS : is the distance between the community and the water source in feet

0.433: conversion factor[4] for feet of H_2O to pounds per square inch

Equation (3) would be applicable only if the water source and the community are at the same elevation. To allow for an elevation difference, the following formulation must be considered:

$$\text{(4)} \qquad EC = (ET - ED) \cdot (0.433)$$

where

EC: difference in elevation between the town site and the water source expressed as pounds per square inch

ET: elevation of town in feet above sea level

ED: elevation of water supply in feet above sea level.

The value of EC can be either positive (the community is higher than the source) or negative (the community is lower than the source). For the positive case, the value of EC is added to LPD to obtain the total system pressure requirements. The negative case can be added to LPD, but if the absolute value of EC is greater than LPD, then a pump is not required. For the case where a pump is required, Equation (5) below applies:

$$\text{(5)} \qquad TLPD = LPD + EC$$

where

TLPD: is the total line pressure drop in pounds per square inch

To ascertain the pump size, the following is used:[5]

$$\text{(6)} \qquad BHP = \frac{Q \cdot TLPD}{(1714)(PE)}$$

where

BHP : brake horsepower of the pump required

PE : the efficiency of the pump (for design purposes, this is often set at 0.8 for both pump and motor)

1714: a constant to maintain unit consistency

Heat Transfer Section

Having considered the pipe size and pump requirements, the heat transfer equations can now be developed. The first consideration is the heat transfer from the air through the insulation, the pipe wall and into the water. For long pipes, which the ones evaluated will be, this heat transfer is in the radial direction of the concentric cylinders. To illustrate the nomenclature, the following illustration is presented:

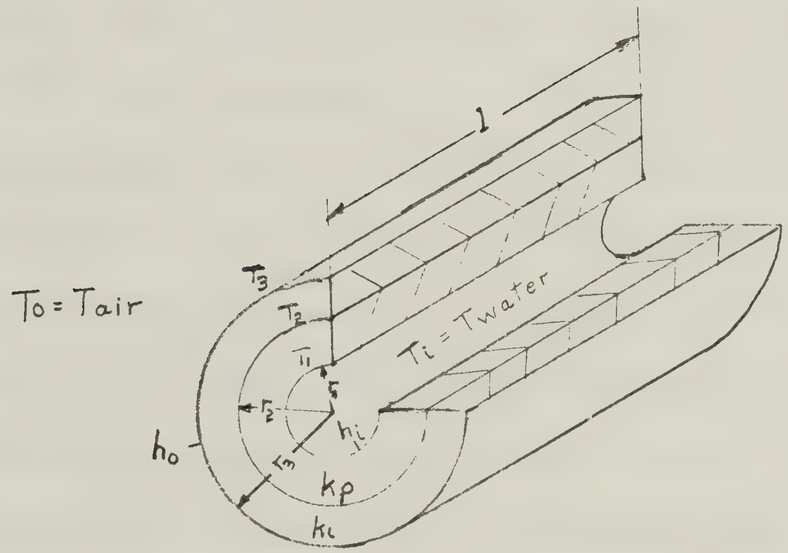

Illustration 1. Illustration of Nomenclature of a Composite Cylinder Wall

For the steady state heat flow condition, the rate of heat flow (q) through each section is the same, and is[6]

(7) $q = 2\pi r_1 l h_i (Ti - T_1)$ for the inner surface

(8) $q = \dfrac{2\pi k p l}{\ln(r_2/r_1)} (Ti - T_1)$ for the inner cylinder

(9) $q = \dfrac{2\pi k_i l}{\ln(r_3/r_2)} (T_2 - T_3)$ for the outer cylinder

(10) $q = 2\pi r_3 l h_0 (T_3 - To)$ for the outer surface

where

q : rate of heat flow in British thermal units per hour (Btu hr^{-1}

r_1 : inside pipe radius in feet (equal to ID/24)

r_2 : outside pipe radius in feet (equal to OD/24)

r_3 : outside radius of the insulation in feet

l : length of pipe being considered in feet

h_i : average unit conductance for the inner surface in units of Btu $hr^{-1} ft^{-2}$ $°F^{-1}$ (for the water)

Ti : bulk temperature of the water in °F

To : bulk temperature of the surrounding air in units of °F

T_1 : temperature of the inner pipe wall in units of °F

T_2 : temperature of the inner insulation area in units of °F

T_3 : temperature of the outer insulation wall in units of °F

kp : thermal conductivity of the pipe in units of Btu hr^{-1} ft^{-1} $°F^{-1}$

ki : thermal conductivity of the insulation in units of Btu hr^{-1} ft^{-1} $°F^{-1}$

h_0 : average unit conductance for the outer surface in units of Btu hr^{-1} ft^{-2} $°F^{-1}$ (for the air)

The temperatures T_1, T_2, and T_3 are not known and not required. By adding the temperature-difference terms and by transposition, T_1, T_2, and T_3 are removed. This gives the following expression of the heat flow from the air to the water:[7]

$$(11) \qquad q = \frac{Ti - To}{\dfrac{1}{2\ r_1 l h_i} + \dfrac{\ln(r_2/r_1)}{2\ kpl} + \dfrac{\ln(r_3/r_2)}{2\ kil} + \dfrac{1}{2\ r_3 l h_0}}$$

To determine an overall heat-transfer coefficient, U, an area on the outer surface will be used as the base area. This area will be denoted as

$$(12) \qquad A_0 = 2\pi r_3 l$$

which gives a heat flow equation as follows, for heat flow from the water to the air:

$$(13) \qquad q = UA_0(Ti - To)$$

Then to put Equation (11) into Equation (13) form, the overall heat transfer coefficient is

$$(14) \qquad U = \frac{1}{\dfrac{r_3}{r_1 h_i} + \dfrac{r_3 \ln(r_2/r_1)}{kp} + \dfrac{r_3 \ln(r_3/r_2)}{ki} + \dfrac{1}{h_0}}$$

where

U : is the overall heat transfer coefficient for the radial conductance, with units of Btu hr^{-1} ft^{-2} $°F^{-1}$

In Equation (11), the terms h_i and h_0 need further correlations. First, consideration must be given to h_i which is expressed by Equation (15) when the constraints of an L/D (length/diameter) ratio of greater

than ten, and a Reynolds number greater than 20,000 are met. The first constraint of L/D greater than ten will always be obtained, as the line length will be long relative to diameter. The velocities employed will always be large enough to give Reynolds numbers exceeding 20,000.

Thus, as the above constraints are met, the following expression for h_i is available:[8]

$$(15) \qquad h_i = 0.026 \left(\frac{ID \cdot G}{12 \cdot u}\right)^{0.8} \left(\frac{Cp \cdot u}{kw}\right)^{0.33} \frac{kw}{ID/12}$$

where

G : mass velocity of the water in units of lbm $hr^{-1}ft^{-2}$

u : viscosity of water at its bulk temperature in units of lbm $ft^{-1}hr^{-1}$

Cp: heat capacity of water at constant pressure on a per unit mass basis, evaluated at the bulk temperature of the water in unts of $ft^2\ hr^{-2}\ °F^{-1}$

kw: thermal conductivity of water at its bulk temperature in units of Btu $hr^{-1}\ ft^{-1}\ °F^{-1}$

0.026: a constant which correlates h_i to experimental data

Note that the expression (ID · G/12·u) is the Reynolds number and (Cp·u/kw) is the Prandtl number in Equation (15).

To determine the mass velocity of water (G) in Equation (15), the following should be used:

$$(16) \qquad G = \frac{0.408 \cdot Q \cdot 3600}{ID^2} \cdot FD$$

where

FD : is the density of water at its bulk temperature in units of $lbm\ ft.^{-3}$

3600: a constant to give time unit consistency

In Equation (16), the term $(0.408 \cdot Q \cdot 3600/ID^2)$ represents the velocity of the water in feet per hour for the actual pipe size used. This will normally be different from the velocity used in Equation (1) because of the use of an actual pipe diameter, as opposed to the original calculated pipe diameter.

The second term from Equation (11) that needs a definition is h_0. This term can be represented as follows:[9]

$$(17) \qquad h_0 = 0.0239 \left(\frac{VA \cdot ODI \cdot}{ua}\right)^{0.805} \frac{ka}{ODI}$$

where

VA : velocity of air normal to pipe in $ft\ hr^{-1}$

ODI : outside diameter of insulation in ft

ua : kinematic viscosity of air at the air's given conditions in units of $ft^2\ hr^{-1}$

ka : thermal conductivity of air at the given ambient temperature in units of $Btu\ hr^{-1}\ ft^{-1}\ °F^{-1}$

Using the above equations, the heat flow, q, from the water to the air for a given length, l, of pipe can be determined. By solving for q, for a length of pipe ten feet or less, a constant temperature for the length of pipe can be assumed. Using this assumption, Equation (13) gives the amount of heat input required to maintain the water at its supply temperature.

The next item that can be found, using the above results, is the amount of temperature drop of the water for the given length of pipe. This can be accomplished with the following formulation:

$$DT = q / (m \cdot Cp) \tag{18}$$

where

DT: bulk temperature drop of the water for the given length of pipe in units of °F

q : is the heat flow out of the water for a given length of pipe in Btu hr^{-1}

m : mass flow rate of the water in lbm hr^{-1}

Cp: heat capacity of water at constant pressure on a per unit mass basis, evaluated at the bulk temperature of the water in units of $ft^2\ hr^{-2}\ {}^{\circ}F^{-1}$ (the same value as used in Equation (15))

Thus, Equation (13) gives the heat loss per unit of pipe length; Equation (18) shows the temperature drop per unit of pipe length. These equations indicate the heat requirements for a tracing system along the length of the pipe to maintain a constant heat flux.

The next consideration is for a system in which the water source temperature is increased, but the line is not heated--only insulated. The approach for these equations will be to develop a system of equations which will show what length of pipe can be used without freeze up--with different levels of insulation on the pipe and different heat inputs at the water source.

To study this system, consider an element of water, of length l, moving through a pipe. Using this element, observe the unsteady state heat transfer around the element, expressed as

(19) $$q = m \cdot Cp \frac{\partial T}{\partial t}$$

$$= FD \cdot \frac{\pi \cdot ID}{48} \cdot 1 \cdot Cp \frac{\partial T}{\partial t}$$

where

$\frac{\partial T}{\partial t}$: change in water bulk temperature (T), per unit time (t), in units of °F for T, and hours for t

Next, take Equation (13) and substitute in for A_0; then equate this to Equation (19), which gives [10]

(20) $$\frac{-\partial T}{\partial t} = C1\ (Ti - To)$$

where

(21) $$C1 = \frac{576 \cdot U \cdot ODI}{FD \cdot ID^2 \cdot Cp}$$

and T is

(22) $$T = To\ (1 - e^{-C1 \cdot t}) + Ti\ e^{-C1 \cdot t}$$

To evaluate the dirivative, it is necessary to consider the temperature of the water at the community. This temperature will be assumed to be just above freezing, and shall be denoted as Tf for the exit water temperature. Considering Ti as the bulk temperature at the water source, with or without heating, then Equation (20) becomes[11]

(23) $$t = \frac{1}{C1}\ \ln\left(\frac{Ti - To}{Tf - To}\right)$$

where

t : is the time in hours the water takes to decrease in temperature from Ti to Tf for the given ambient temperature, To.

Using this information, the distance the water can be moved without freezing is

(24) $$PL = t \cdot VWP \cdot 3600$$

where

PL : is the length of pipe in feet that can be used without the water freezing

VWP : is the mean velocity of the water in the pipe in ft sec^{-1} (note: $VWP = 0.408\ Q/ID^2$ and 3600 sec hr^{-1} as t in hours)

3600: constant to maintain unit consistency

By substituting from Equations (21) and (23), Equation (24) can be rewritten as follows:

(25) $$PL = \frac{FD \cdot ID^2 \cdot Cp}{576 \cdot U \cdot ODI} \ln\left(\frac{Ti - To}{Tf - To}\right) \cdot VWP \cdot 3600$$

Thus, Equation (25) gives the maximum pipe length that can be used for a given installation. Using this equation, the insulation thickness could be varied--which changes both U and ODI--and this effect studied. Alternately, the water source temperature, Ti, could be varied and its effect studied for different insulation types and thicknesses.

The above presents the complete set of physical equations necessary to solve the physical problems related to freeze up prevention of an overland water pipeline in a cold environment. Chapter III will illustrate, with an example, the usage of the above equations.

CHAPTER III

QUANTITATIVE EXAMPLE

To illustrate the use of the equations developed in Chapter II, and to ascertain the significance of the various quantities, a numerical example is now presented. The results from the example will show which equations can be simplified, and which ones can be removed.

First, the basic data to be used for the example includes:

1. Community size and water usage, which could be either of the following:

number of residents	usage as imperial gallons per person per day	net daily usage in imperial gallons
3,000	20	60,000
2,000	30	60,000

2. Community and water source at the same elevation
3. Community is two miles from the source
4. Two inches of styrofoam insulation
5. Water source temperature of 36°F
6. Ambient temperature of -50°F, and a wind velocity of 50 miles per hour
7. An initial water velocity in the pipe of 7 feet per second

Fluid Flow

With the above basic data, the equations can now be used to find various quantities. Equation (1) will determine the theoretical pipe diameter. Equation (1) is

(1) $$D = ((0.408 \cdot Q) \cdot VW^{-1})^{0.5}$$

where

$$Q = 60{,}000\ \frac{\text{imp gal}}{\text{day}} \cdot \frac{1.201\ \text{US gal}}{\text{imp gal}} \cdot \frac{\text{day}}{24\ \text{hr}} \cdot \frac{\text{hr}}{60\ \text{min}} = 50.04\ \text{usgm}$$

$VW = 7$ ft sec^{-1}

This gives D = 1.71 inches. With this value, the following values for a commercial pipe can be found on Table I (page 6).

ID = 2.067 inches and OD = 2.385 inches. For the remainder of the example development, these actual diameters are used to give the most realistic evaluations.

Now it is possible to observe the line pressure drop per 100 feet of pipe which is expressed as

(2) $$f = 0.2083 \left(\frac{100}{C}\right)^{1.85} \cdot \frac{Q^{1.85}}{(ID)^{4.8655}}$$

where

Q = 50.04 usgpm

C = 100 assumed for design purposes, as the pipe is steel

ID = 2.067 inches

which gives f = 8.48 feet of water per 100 feet of pipe

The example requires a line two miles long. The pressure drop for the total line is found using Equation (3) as follows:

(3) $$LPD = f \cdot 0.433 \cdot sp.gr \cdot DS/100$$

where

f = 8.48 feet of water per 100 feet of pipe

sp.gr = 1 for water at the given conditions

DS = 2 miles · 5280 feet/mile = 10560 feet

which gives LPD = 387.7 pounds per square inch.

As ET = ED, then EC from Equation (4) is equal to zero. Therefore, TLPD of Equation (5) is equal to 387.7 psi. To determine the pump horsepower, a design efficiency of 80 per cent will be used (this value is normally used for design purposes). Equation (6) determines the pump horsepower.

$$BHP = \frac{Q \cdot TLPD}{(1714)(eff)} \tag{6}$$

where

Q = 50.04 usgpm

TLPD = 387.7 psi

eff = 0.80

which gives 14.1 horsepower.

<u>Heat Transfer</u>

Looking now at the heat transfer equations, the parameters can now be calculated. The value of U is required first. To obtain U, Equations (15), (16) and (17) must be solved. First Equation (16) must be solved for the water mass velocity, which is

$$G = \frac{0.408 \cdot Q \cdot 3600}{ID^2} \cdot FD \tag{16}$$

where

Q = 50.05 usgpm

ID = 2.067 inches

FD = 62.4 lbm ft^{-3}

which gives G = 1,073,454 lbm $hr^{-1}ft^{-2}$

This value can be used in Equation (15) to determine h_i. The solution is

(15) $$h_i = 0.026 \left(\frac{ID \cdot G}{12 \cdot u}\right)^{0.8} \cdot \left(\frac{Cp\ u}{kw}\right)^{0.33} \frac{kw}{ID/12}$$

where

ID = 2.067 inches

G = 1,073,454 lbm hr^{-1} ft^{-2}

u = 0.00112 lbm ft^{-1} sec^{-1} · 3600 sec hr^{-1} = 3.88 lbm ft^{-1} hr^{-1}

(See footnote 12)

Cp = 1.01 Btu lbm^{-1} $°F^{-1}$ (See footnote 13)

kw = 0.323 Btu hr^{-1} ft^{-1} $°F^{-1}$

which gives h_i = 611.7 Btu hr^{-1} ft^{-2} $°F^{-1}$ and a Reynolds number of 47,655.

Similarly, the value of h_0 from Equation (17) is

(17) $$h_0 = 0.0239 \left(\frac{VA \cdot ODI}{ua}\right)^{0.805} \frac{ka}{ODI}$$

where

VA = 50 miles hr^{-1} · 5280 ft $mile^{-1}$ = 264000 ft hr^{-1}

ODI = (2.375 + 4)/12 = 0.5312 ft

ua = 0.000107 · 3600 = 0.3850 lbm ft^{-1} $°F^{-1}$ (See footnote 15)

ka = 0.0122 Btu hr^{-1} ft^{-1} $°F^{-1}$ (See footnote 16)

which gives h_0 = 16.5 Btu hr^{-1} ft^{-2} °F.

Sufficient information is now supplied to determine the overall heat transfer coefficient, U. There are four distinct groups of parameters in the definition of U. Each group represents the resistance to heat flow for a particular part of the system. Reading Equation (15) from left to right, the four groups relate to Equations (7), (8), (9) and (10), respectively, as does the heat flow moving through the

resistances radially outward from the water. To observe the magnitude of the resistance of each component, each term will be evaluated separately. The first term to consider is the resistance of the water next to the pipe wall. The resistance is represented by

$$r_3 / (r_1 h_i)$$

where

$r_3 = 6.375/24 = 0.2656$ feet

$r_1 = 2.067/24 = 0.0861$ feet

$h_i = 611.7$ Btu hr^{-1} ft^{-2} °F^{-1}

This gives a heat flow resistance of 0.0050 hr °F Btu^{-1}. Moving outward, the next heat transfer resistance encountered is the pipe wall. The resistance is represented by

$$kp / (r_3 \ln (r_2/r_1))$$

where

$r_3 = 0.2656$ feet

$r_2 = 2.375/24 = 0.0989$ feet

$r_1 = 2.067/24 = 0.0861$ feet

$kp = 26.5$ Btu hr^{-1} ft^{-1} °F^{-1}(See footnotes 17)

This gives a heat flow resistance of 0.0014 hr °F Btu^{-1} for the pipe wall. The next resistance encountered is the two inches of styrofoam insulation. The resistance for the insulation is represented by

$$ki / (r_3 \ln(r_3/r_2))$$

where

$r_3 = 0.2656$ feet $\quad r_2 = 0.0989$ feet $\quad r_1 = 0.0861$ feet

$ki = 0.24$ Btu hr^{-1} ft^{-1} $°F^{-1}$ (See footnote 18)

This gives a heat flow resistance of 1.093 hr °F Btu^{-1} for the insulation. It should be noted that this value is the largest thermal resistance. The final resistance encountered is the outside air film. This resistance is represented by

$$1 / h_0$$

where

$h_0 = 16.5$ Btu hr^{-1} ft^{-2} $°F^{-1}$

which gives a value of 0.060 hr °F Btu^{-1} for the heat flow resistance of the air film.

Using the above quantities in Equation (14), a value of 0.86 Btu hr^{-1} ft^{-2} $°F^{-1}$ is obtained for U. Observing the contribution of each resistance to the overall resistance, the only significant term is for the insulation. Due to this order of magnitude difference between the insulation term and the other terms, Equation (15) will be simplified to contain only the insulation term. This gives the following equation for the model:

$$U \simeq ki \,/\, (r_3 \ln (r_3/r_2)) \tag{26}$$

This eliminates the need to solve Equations (15), (16) and (17) for the model. Using the value of U and substituting Equation (12) into Equation (13), the heat transfer for a given length of pipe can be solved. This represents the amount of heat required to maintain a constant heat flux along the line. For this case, the unit length chosen is ten feet. The modified equation is

(27) $$q = U2\pi r_3 l\ (Ti - To)$$

where

$U = 0.86$ Btu hr^{-1} ft^{-2} $°F^{-1}$

$r_3 = 0.2656$ feet

$l = 10$ feet

$Ti = 36°F$

$To = -50°F$

which gives $q = 1234$ Btu hr^{-1} for ten feet of pipe. This value indicates the heat input required for tracing along the length of the line to maintain the water at the bulk temperature, Ti. This information can be employed for evaluation of this type of freeze up prevention system--for different thicknesses of insulation--which changes the U value and, hence, the q value.

Corresponding to this is the temperature drop for the ten feet of pipe if q is allowed to dissipate. This is found using Equation (18), which for the example is

(18) $$DT = q\ /\ (m \cdot Cp)$$

where

$q = 1234$ Btu hr^{-1} per ten feet of pipe

$$m = 60{,}000\ \frac{\text{imp gal}}{\text{day}} \cdot \frac{\text{day}}{24\ \text{hr}} \cdot 10.02\ \frac{\text{lbm}}{\text{imp gal}} = 25050\ \frac{\text{lbm}}{\text{hr}}$$

$Cp = 1.01$ Btu lbm^{-1} $°F^{-1}$ (See footnote 19)

which gives $DT = 0.0488$ °F per 10 feet of pipe.

Complementing this heat dissipation temperature drop term for a short given length of pipe, is Equation (25), which gives the total length of pipe. This can be used to observe the length of pipe which

is possible with differing temperature-driving forces. Equation (25) gives these results for the example:

$$(25) \qquad PL = \frac{FD \cdot ID^2 \cdot Cp}{576 \cdot U \cdot ODI} \ln\left(\frac{Ti - To}{Tf - To}\right) \cdot VWP \cdot 3600$$

where

$FD = 62.4\ lbm\ ft^{-3}$

$ID = 2.067$ inches

$Cp = 1.01\ Btu\ lbm^{-1}\ {}^\circ F^{-1}$

$U = 0.86\ Btu\ hr^{-1}\ ft^{-2}\ {}^\circ F^{-1}$

$ODI = 0.5312$ feet

$Ti = 36^\circ F$

$To = -50^\circ F$

$Tf = 33^\circ F$

$VWP = 0.408 \cdot 50.04 / (2.067 \cdot 2.067) = 4.8\ ft\ sec^{-1}$

This gives PL = 618.9 feet. As shown, Equation (25) is given in terms of velocity, but by substituting for VWP, it can be rewritten in terms of Q, as follows:[20]

$$(28) \qquad PL = 2.55\ \frac{FD \cdot Cp \cdot Q}{U \cdot ODI} \ln\left(\frac{Ti - To}{Tf - To}\right)$$

This equation can be used to determine the overall pipe length--which does not require any heat input along the line--for a given set of temperatures. Or, by specifying the pipe length, Ti, the source water temperature could be solved for. By solving for Ti, the amount of heat required at the source to enable the water to reach the community without any additional heat along the line, is determined.

The example shows that the insulation type and thickness control the heat transfer to the water. By using this information, various insulation alternatives can be evaluated with different types of heat input systems--either tracing along the line and/or heating of the source water. The above equations supply the necessary physical values for these evaluations. The equations required for evaluation are assembled in Chapter IV, where the same example is further used to generate groups of alternatives. To establish a method of evaluation, the associated cost formulations are related to the required physical quantities in Chapter IV.

CHAPTER IV

MODEL DEVELOPMENT

Expanding on Chapters II and III, this chapter will develop a model which can be used to study the various physical systems, as well as their associated costs. The equations required will be assembled into four related groups. The group structure will be as follows:

Group 1: equations for pipe and pump size

Group 2: equations for heat transfer for a short length of pipe

Group 3: equations for heat transfer considering total pipe length and variations in source water temperature

Group 4: cost equations related to Groups 1, 2 and 3

Initially, Group 1 will be discussed, giving its make-up. The discussion of Group II will present an example of different insulation thicknesses--showing their effects on U, q and DT. The Group 3 presentation gives two examples: the first illustrating the effect that varying the insulation thickness and the source water temperature has on the unheated pipe length, and the second illustrating the necessary source water temperature for different insulation thicknesses to transport the water a given pipe length. Also shown is the amount of heat required to raise the water temperature to the different indicated temperatures. The Group 4 development will conclude the discussion by giving cost equations for each different system. This will enable the user to obtain a variety of results, depending on his specific requirements.

Group 1 will consist of Equations (1) to (6), inclusive. The normal input values specific to a given community are water requirement, community elevation, and the water source elevation. A decision-variable input to Group 1 is the water velocity. The velocity term coupled with the community values starts the model, as the model is data driven. The output from this group is comprised of the pipe diameter, the line pressure drop, and the pump horsepower. The pipe diameter is an input for Group 2; the pump horsepower is an input for Group 4.

Group 2 will consist of the heat transfer equations for the constant line temperature case, comprising Equations (18), (26) and (27). The input values employed specific to a community are the water requirements, the water source temperature, and the coldest ambient air temperature. Decision variables related to the physical system are the insulation thickness (represented as the overall outside diameter), the thermal conductivity of the insulation, and a unit length of pipe for design considerations. To illustrate the effect of varying the insulation thickness, the following example is given, which is based on the example previously used in Chapter III.

Using insulation thicknesses of 2 inches, 3 inches and 4 inches gives the results shown in Table II.

Table II shows the effect on U, q and DT that different thicknesses of insulation have. The last column gives the amount of heat (q) that is necessary to maintain the water at its source temperature by adding heat evenly along the length of the line. Using this information the Group 4 equations will present the cost trade-off comparison of the capital cost of more insulation versus the cost of adding heat to the line.

TABLE II

EFFECT OF INSULATION THICKNESS ON HEAT INPUT FOR THE LENGTH OF THE PIPE

Water Source Temp °F	Insulation Thickness Inches	U (Btu $hr^{-1}ft^{-2}$ $°F^{-1}$) Equation (27)	q for 10 feet Btu/hr Equation (26)	DT for 10 feet °F Equation (18)	Total q for 2 Miles Btu/hr
36°F	2	0.915	1313	0.0519	1,386,528
	3	0.507	956	0.0378	1,009,536
	4	0.377	881	0.0348	930,336

(The q is input equally along the length of the line)

Group 3 employs the heat transfer equations (27) and (28) which consider the effect of heating the water at the source with no additional heat along the length of the line. The input values specific to the community are the water requirement, the coldest ambient air temperature, and the source water temperature. The decision variables are the temperature of the water at the community, the thickness of the insulation, and the thermal conductivity of the insulation. The heating of the water from its supply temperature to a higher temperature, Ti, before it enters the pipeline, requires a heat flow input. This can be found using equation (18) in the following form:

$$qs = m \cdot Cp \cdot (Ti - Ts) \quad (29)$$

where

qs: is the heat input to raise the water source temperature to a higher temperature in Btu hr^{-1}

m : mass flow rate of the water in lbm hr^{-1}

Cp: heat capacity of water at constant pressure on a per unit mass, evaluated at the bulk temperature of the water, in units of $ft^2\ hr^{-2}\ °F^{-1}$

Ti: temperature of the water entering the pipeline in °F

Ts: temperature of water source before heating, in °F

Using Equations (29), (28) and (27), an example showing the effect of varying the source temperature and the insulation thickness is shown in Table III.

For each case there is a large gain in distance where the insulation thickness is increased from two inches to three inches, and a small gain from three inches to four inches. The other way of using Equations (28) and (29) is to use the given pipe length, which for the example is two miles (11560 feet), and to solve for the temperature to which the source water needs to be heated. With this temperature, Equation (29) can be solved to ascertain the heat input requirements. For the example given--using two, three, and four inches of insulation--the following results are obtained in delivering the water to the community at 33°F (Table IV).

Thus, for the example any of the above systems would supply the community with water, but each system would have a different cost. To evaluate the costs, the following group of cost equations is presented.

TABLE III

EFFECT OF WATER SOURCE TEMPERATURE

AND INSULATION THICKNESS ON UNHEATED PIPE LENGTH

Water Source Temp °F	Insulation Thickness Inches	U (Btu hr^{-1} ft^{-2} $°F^{-1}$ Equation (27)	PL (ft) without q added along pipe Equation (28)	qs (Btu hr^{-1}) heat added at source Equation (29)
36	2	0.915	587	0
	3	0.507	807	0
	4	0.377	875	0
50	2	0.915	3083	354207
	3	0.507	4235	354207
	4	0.377	4597	354207
70	2	0.915	6100	860217
	3	0.507	8389	860217
	4	0.377	9095	860217
100	2	0.915	9792	1619232
	3	0.507	13452	1619232
	4	0.377	14600	1619232

Note: temperature of water arriving at community is 33°F for the above. PL is the pipe length the water can travel before its temperature drops to 33°F.

TABLE IV

SOURCE WATER HEATING REQUIREMENTS TO DELIVER THE WATER TO THE COMMUNITY WITH DIFFERENT AMOUNTS OF INSULATION

Insulation Thickness Inches	U (Btu hr^{-1} ft^{-2} $°F^{-1}$ Equation (27)	Temperature Water Heated To, Ti °F Equation (28)	qs (Btu hr^{-1} Heat Added to Bring Source Water to Ti Equation (29)
2	0.915	107	1,796,336
3	0.507	82	1,163,823
4	0.377	77	1,037,320

Note: PL = 2 miles for the above

The cost equations for Group 4 will be constructed to consider first the operating costs and the capital costs. Secondly, these values will be put in a present value format to provide the net present value of the cost for n years of operation.

Two sets of operating and capital costs will be developed. The first set uses the results from Group 1 and Group 2 above. These equations will reflect the cost of tracing along the line. The second set of cost equations will use the output from Group 1 and Group 3, allowing for the cost of heating the source water, but having no tracing cost term. The capital cost for the traced system is

$$CCT = DS\ (PC + TC + IC)(1 + ITC) + (PUC + DRC)(1 + IPC) \tag{30}$$

where

CCT: capital cost in dollars for the traced system

PC : pipe cost on a dollars-per-foot basis (note: this varies with size and type)

TC : tracing cost on a dollars-per-foot basis

IC : insulation cost on a dollars-per-foot basis (note: this varies with thickness and type)

ITC: installation cost as a decimal percentage of the equipment cost

PUC: pump cost in dollars

DRC: driver cost in dollars

IPC: pump and driver installation cost as a decimal of the pump and driver cost

The associated operating cost for the traced system is

$$VCT = OHY\ (BHP \cdot EHP + HC \cdot Tq + OCT + SCT) \tag{31}$$

where

VCT: yearly operating cost for a traced pipeline system in units of dollars per year

OHY: operational hours per year (normally used 24 hours/day · 365 days/year = 8760 hours per year)

EHP: energy cost, expressed as dollars per horsepower per hour

HC : cost of energy expressed as dollars per Btu

Tq : this is the total heat input along the line in Btu per hour (note: this is q per length 1 times DS per 1)

OCT: hourly operator cost, in dollars per hour, for a traced system

SCT: service cost on a dollar per hour basis for a traced system

Using the above capital and variable cost equations, the net present value cost equation for n years of operation is

(32) $$PVTS = CCT + VCT\ (1 - (1 + IR)^{-n} / IR)$$

where

PVTS: is the present value in dollars of the traced system for n years

IR : the cost of capital expressed as a decimal percentage on a yearly basis

n : number of years for the evaluation

The above gives the complete set of cost formulas for a traced and insulated overland water pipe system. The next set of equations will give the same information for a system not traced, but with the supply water heated. The capital cost equation is

(33) $$CCSH = DS\ (PC + IC)\ (1 + ISH) + (PCV + DRC)\ (1 + IPC) + (HC \cdot qs)\ (1 + ICH)$$

where

CCSH: capital cost, in dollars, for the non-traced system

ISH : installation cost of the pipe and insulation, expressed as a decimal of the equipment cost

HC : heater cost on a dollars per Btu per hour basis

qs : heat flow into the source water in Btu per hour

ICH : installation cost of the heater, expressed as a decimal of the heater cost

The variable cost for the water source heated system is expressed as

$$(34) \quad VCSH = OHY\ (BHP \cdot EHP + HC \cdot qs + OCSH + SCSH)$$

where

VCSH: yearly operating cost for a pipeline system, with the water heated before entering the line in dollars per year

OCSH: operator cost in dollars per hour for a heated water supply system

SCSH: service cost in dollars per hour, for a heated water supply system

With the above capital and operating cost equations, the following net present value (cost) equation for n years operation of a heated water source system can be expressed as

$$(35) \quad PVSH = CCSH + VCSH\ (1 - (1 + IR)^{-n}\ /\ IR)$$

where

PVSH: is the present value, in dollars, of the heated water supply system for n years

This is, in conclusion, the final equation necessary to complete the model. The preceding displayed how various physical design parameters could be varied to give different physical configurations of systems all supplying the same overall service. To evaluate which system is the most economical for a given situation, the present value cost equations are supplied. Thus, the model takes a group of specific input parameters for a community, and with a group of decision variables supplied by the user, provides an economic assessment of alternative pipeline designs.

CHAPTER V

CONCLUSION

The foregoing examination of the problem of transporting water using an insulated overland pipeline, illustrates the significance of various design parameters. Their significance appears both in the physical configurations determined, and in the relative costs of the various alternatives.

Two methods of freeze up prevention were considered--a line traced along its length, or heating of the water before it entered the line. With each of these methods, the type and amount of insulation is one of the prime decision variables. The use of the model is not constrained by only presenting the two basic methods of heat input to the system, as once the heat input requirement is known, a large variety of heat supply systems can be evaluated. For example, if it appears that tracing is the best method, then upon consideration of the type of energy source available, the various types of tracing--such as hot water recirculation, steam tracing, electric tracing or some other type--can be evaluated on a cost basis. To design such a tracing system, the main information requirement is how much heat is to be input per unit of pipe length. This information is supplied by the model.

Similarly the alternate method of freeze up prevention--heating the water before it enters the line--can be accomplished using a variety of equipment, such as a direct fired furnace, heat exchanger or an electrical heater. The prime information necessary

for the design of this type of heater, is the heat capacity which the model supplies. Thus, although there are numerous methods of supplying the heat to the system, the main requirement is the quantity of heat and where it should be input. With this information, it is then an equipment selection problem, not a pipeline design problem.

Thus, the model provides the necessary physical information for the design of the basic components of an overland water pipeline system in northern Canada. With the foregoing physical design information, the model provides the cost structure of alternative designs, permitting the evaluation of the alternatives.

TABLE V

SYMBOLS USED AND THEIR LOCATION

Symbol	Page	Symbol	Page	Symbol	Page
D	5	kp	10	TC	32
Q	5	ki	10	IC	32
VW	5	h_0	10	ITC	32
OD	6	Ao	11	PUC	32
f	7	U	11	DRC	32
C	7	G	12	IPC	32
ID	7	u	12	VCT	32
LPD	7	Cp	12	OHY	32
DS	8	kw	12	EHP	32
EC	8	FD	12	HC	32
ET	8	VA	13	Tq	32
ED	8	ODI	13	OCT	32
TLPD	8	ua	13	SCT	32
BHP	8	ka	13	PVTS	33
PE	8	DT	14	IR	33
q	10	m	14	n	33
r_1	10	T	15	CCSH	33
r_2	10	t	15	ISH	33
r_3	10	Cl	15	HC	33
l	10	Tf	15	ICH	33
h_i	10	PL	16	VCSH	34
Ti	10	VWP	16	OCSH	34
To	10	qs	28	SCSH	34
T_1	10	Ts	29	PVSH	34
T_2	10	CCT	32		
T_3	10	PC	32		

FOOTNOTES

[1]Engineering Division, Crane Co., *Flow of Fluids*, p. 3-2. Throughout the thesis, the nomenclature of the sources indicated may be changed to provide consistency.

[2]*Ibid.*, p. B-16.

[3]G. V. Shaw and A. Loomis, eds., *Cameron Hydraulic Data*, p. 27.

[4]*Ibid.*, p. 27.

[5]*Ibid.*, p. 12.

[6]Frank Kreith, *Principles of Heat Transfer*, p. 42.

[7]*Ibid.*, pp. 42-3. Equations (11), (12), (13) and (14) are patterned after the Kreith development.

[8]B. Bird, W. Stewart, and E. Lightfoot, *Transport Phenomena*, p. 399. Note: μ_b/μ_0 was removed from Equation (13, 2-16) because this quantity equals one in this type of environment.

The Reynolds number is a dimensionless group which indicates whether the flow of fluid in a pipe is turbulent. For Reynolds numbers greater than 20,000 flow is considered turbulent.

The Prandlt is a dimensionless group used in forced convective heat transfer studies.

[9]Kreith, p. 468. The coefficients for the equation are for the case of a high Reynolds number, as the design will be done at maximum wind velocities, normal to a relatively large pipe OD when the insulation is considered.

[10]Equating Equations (13) and (19) and allowing for the different heat flow directions results in

$$- FD \; \frac{\pi}{4} \; \frac{ID}{12}^{2} \cdot 1 \cdot Cp \frac{\partial T}{\partial t} = U \cdot ODI \cdot 1 \cdot (Ti - To)$$

$$- \frac{\partial T}{\partial t} = \frac{576 \cdot U \cdot ODI}{FD \cdot ID^2 \cdot Cp} \cdot (Ti - To)$$

[11]Solve Equation (22) as follows:

$$- \frac{\partial T}{\partial t} = C1 \; (Ti - To)$$

$$\frac{\partial T}{\partial t} + C1 \cdot Ti = C1 \cdot To$$

$$T \; e^{C1 \cdot t} = To \; e^{C1 \cdot t} + C2$$

$$T = To + C2\ e^{-C1 \cdot t}$$

Solve for C2:

if $t = 0$, $T = Ti$

therefore, $Ti = To + C2$

$$T = To + (To - Ti)e^{-C1 \cdot t}$$

$$T = To\ (1 - e^{-C1 \cdot t}) + Ti\ e^{-C1 \cdot t}$$

Let Tf denote the water outlet bulk temperature and t be the time it takes the water to reach Tf from Ti.

$$T = \frac{1}{C1} \cdot \ln \frac{Ti - To}{Tf - To}$$

[12]Kreith, p. 638.

[13]*Ibid.*

[14]*Ibid.*

[15]*Ibid.*, p. 636.

[16]*Ibid.*

[17]*Ibid.*, 638.

[18]This is the ki for styrofoam as supplied by J-K Campbell and Associates Ltd., Edmonton, Alberta

[19]Kreith, p. 638.

[20]
$$PL = \frac{FD \cdot ID^2 \cdot Cp \ln}{576 \cdot U \cdot ODI} \quad \frac{Ti - To}{Tf - To} \cdot \frac{0}{ID^2} \cdot 3600$$

Note: ID^2 cancel and $3600/576 = 2.55$.

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